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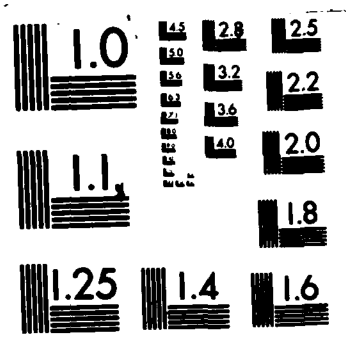

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AD A138368

SHIP MOTION GENERATOR UPGRADE STUDY

William H. Muzzy III

September 1983

NAVAL BIODYNAMICS LABORATORY
New Orleans, Louisiana

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REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER NBDL - 83R012	2. GOVT ACCESSION NO. AD-A138 368	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) SHIP MOTION GENERATOR UPGRADE STUDY		5. TYPE OF REPORT & PERIOD COVERED RESEARCH REPORT
		6. PERFORMING ORG. REPORT NUMBER NBDL - 83R012
7. AUTHOR(s) WILLIAM H. MUZZY III		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS NAVAL BIODYNAMICS LABORATORY BOX 29407 NEW ORLEANS, LA 70189		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS PROJECT #F58524 TASK AREA MZF58534026 WORK UNIT MF58.524-026-30
11. CONTROLLING OFFICE NAME AND ADDRESS		12. REPORT DATE SEP 1983
		13. NUMBER OF PAGES 19
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS (of this report) UNCLASSIFIED
		15a. DECLASSIFICATION DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE, DISTRIBUTION UNLIMITED

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

SHIP MOTION, HEAVE DRIVE, PITCH, ROLL, STROKE, PISTON, HYDRAULIC PRESSURE, ACCELERATION, VELOCITY, HYDRAULIC OIL, FREQUENCY

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A study was undertaken to determine the specifications for a piston that would bring the Ship Motion Generator back to the original heave displacement of ±11 feet, reduce piston whip, decrease the lower overtravel buffer deceleration and retain the 17 ft. per second maximum velocity. Recommendations were made to modify the heave drive system and tower structure.

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S/N 0102-LF-014-6601

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

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SHIP MOTION GENERATOR UPGRADE STUDY

William H. Muzzy III

September 1983

Bureau of Medicine and Surgery Work Unit No. MF5852402630

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New Orleans, Louisiana

Summary

The Ship Motion Generator (SMG) was disassembled and removed from its original installation site at Human Factors Research, Inc., in Santa Barbara, California in March 1977. It was shipped to the Navy's Civil Engineering Laboratory at Port Hueneme, California for cleaning and painting. Later it was shipped to Naval Biodynamics Laboratory, New Orleans, Louisiana for installation in Test Cell #3, Building 420. The "downtime" afforded a chance to upgrade the device; modifications were already necessary in adapting it to its new site.

Changes were made that allowed the device to attain its original heave stroke of ± 11 feet while retaining the upgraded 17 ft/sec maximum velocity with a 2:1 safety factor. The lower overtravel buffer length was increased to lower the deceleration from 6.5G to 2G.

Modifications and changes to the tower structure and heave drive system are summarized below.

1. Increase the tower height by 9' (to 41'-4").

2. Install a new lower overtravel buffer with a 3'0" stroke length.

3. Install an 8" diameter Schedule 40 steel pipe 40 feet long as the new heave drive casing.

4. Fabricate a new bearing/seal to fit the heave drive casing with an ID of 4.5 inches.

5. Install a new heave drive piston 4.5 inches in diameter and 30 feet long.

6. Change the heave drive supply line from a 4-inch ID to a 6-inch ID and the return line from a 6-inch ID to an 8-inch ID.

7. Change the heave drive piping configuration to align the safety valve T-fitting and control valve in order to decrease back pressure.

8. Change the safety valve from a globe valve design to a ball valve design and increase the size from 4 inches to 6 inches.

9. Increase the control valve size from 4 inches to 6 inches.

10. Fabricate and install a duplicate heave drive power supply to achieve the 17 ft/sec maximum velocity with the 4.5-inch diameter piston.

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SHIP MOTION GENERATOR UPGRADE STUDY

HISTORY

In March of 1969, Human Factors Research, Incorporated (HFR), in Santa Barbara, California, completed construction of a machine capable of simulating seacraft motions in wind and water conditions of up to Sea State Five with three degrees of freedom (heave, pitch, and roll). The work was sponsored by the Aeronautics Programs Branch, Naval Applications and Analysis Division of the Office of Naval Research. The device was a U.S. Navy facility, maintained and operated by HFR, for the purpose of conducting basic and applied research on human reactions in moving environments. It had become known widely as the ONR/HFR Motion Generator.

The Motion Generation was originally designed to Navy specifications that would allow "barge-like" motion simulations. As it happened, the performance of the completed device considerably exceeded the design goals.

The impact of unconventional hull design for seacraft of the future engendered a new requirement for extensive motion simulation research. The motion environment anticipated aboard such craft would be unlike that commonly found in ships of conventional design. In particular, heave acceleration levels would be greater, particularly at frequencies between 0.5 Hz and about 2.0 Hz. Habitability problems may occur which could conceivably limit the types of operations performed in those craft or necessitate the redesign of shipboard life-support and working-area systems.

The original ONR/HFR Motion Generator came close to being a suitable device for simulating motions of unconventionally hulled seacraft, but fell short of providing the heave motion spectrum anticipated under the more severe combinations of ship speed and sea state.

The Surface Effect Ships Project Office (SESPO PMS 304) of the Naval Sea Systems Command let a contract in 1974 for the purpose of upgrading the Motion Generator's system performance. Modifications and testing were completed in the spring of 1975 and man rated in July 1975,

Most structural elements of the ONR/HFR Motion Generator remained unchanged in the upgraded version. The principal changes occurred in command signal processing electronics, the heave drive system, and the various safety systems.

The original tower was 31 feet high. With carriage travel of 22 feet, the maximum wave amplitude was ± 11 feet, centered at the 16-foot mark on the tower. The upgraded carriage travel was reduced to 20 feet to provide a maximum wave amplitude of ± 10 feet, increase heave velocity to 17 ft/sec and allow additional space for upgraded top and bottom buffers.

A STUDY WAS UNDERTAKEN to determine the specifications for a piston that would bring the Ship Motion Generator back to the original heave displacement of ± 11 feet, reduce piston whip, decrease the lower overtravel buffer deceleration and retain the 17-foot per second maximum velocity.

PISTON LENGTH

Prior to analyzing piston stresses, the piston length had to be sized to accommodate the 22-foot stroke as well as the upper and lower overtravel. Stopping the carriage from an upward overtravel condition with a constant deceleration is given by the classic formula

$$\text{distance} = \frac{\Delta v^2}{2a} \tag{1}$$

where Δv is the change in velocity in ft/sec and a is the average deceleration in ft/sec^2.

Assuming a maximum velocity of 17 ft/sec at the top of the stroke (+11 ft) with the heave drive power off and an average deceleration of 64.24 ft/sec^2 (2G), then the distance required to bring the carriage to a full stop is 2.24 ft (27 inches) which is the same as is presently available on the SMG tower with the spring buffer.

In arresting the downward motion it is assumed that the carriage free falls from the top of the fully compressed buffer. Using (1) again with a 1G acceleration, the velocity at lower buffer contact is 39.5 ft/sec. To decelerate the carriage to a full stop with the same requirement of 2G would require 12.14 ft. This is unacceptable as it would locate the -11 ft. mark over 23 ft. above the floor. A more practical maximum downward velocity is 20 ft/sec. Note: Alfred Z. Boyajian and Wilton A. Stewart assumed a 17 ft/sec maximum downward velocity for their lower buffer calculations while drop tests from the highest attainable position on the tower resulted in an 18.7 ft/sec terminal velocity and an ensuing 6.5G deceleration [1]. Therefore, recalculating the deceleration distance with 20 ft/sec and 2G results in a buffer stroke length of 37.4 inches. A commercially available single stage oil buffer with a stroke length of 36 inches, and an outside diameter of 6 inches, was installed to replace the original lower buffer.

With the upper and lower buffer deceleration distances fixed at 2.24 ft. and 3.0 ft., the tower height was calculated

a.	Distance from floor of cab to lowest point on wheels	96.00"
b.	Mandatory clearance from wheels to lower base	3.50"
c.	Tower base	1.00"
d.	Grout between tower base and foundation	2.00"
e.	Minimum cab floor height above ground without buffer	102.00"
f.	Buffer stroke distance	36.0"

g.	Buffer housing height	57.0"
h.	Total buffer height - unstroked	93.0"
i.	Distance from floor of cab to piston interface	54.75"
j.	Minimum cab floor height above ground with buffer stroked (g + i > e)	111.75"

In order to have a convenient height above ground for the control/computer room that would allow a serviceable work area beneath it, the cab floor height was located 132.0" (11'-0") above the test cell floor. With an increased lower buffer length, a parked cab floor height of 11 feet, and a 22-foot maximum heave displacement, the tower height was increased by 108 inches (from its original 32'-4" to 41'-4"). Two bays similar to those on the existing structure were fabricated and welded to the tower by contractor personnel.

The piston length was sized for the new motion requirements

Heave displacement ± 11'-0"	22'-0"
Upper overtravel buffer	2'-3"
Lower overtravel buffer	3'-0"
Distance from piston casing to top of buffer housing	1'-0"
Piston remaining inside casing	1'-0"
Required piston length	29'-3"

As a point of comparison, the original 24-foot, 3-1/2 -inch diameter piston would have ± 8'-8.5" displacement if it were installed in place of the recommended one.

PISTON DIAMETER

Boyajian and Stewart calculated a maximum allowable load on the original piston as 14,130 pounds and a maximum allowable compressive stress of 4520 psi [2]. This resulted in a safety factor of 1.47 over the expected peak compression loads and stresses.

In the redesign, a safety factor of 2:1 over expected peak loads was considered as minimum. With the roll and pitch performance criteria remaining the same, the new heave performance goals are as follows:

Heave	Original	Upgraded	NBDL Installed
Displacement	±11 feet	±10 feet	±11 feet
Max. Velocity	± 8.7 ft/sec	±16.6 ft/sec	±17 ft/sec
Max. Acc.	± .35G	+ 1.0G - 0.8G	+ 1.0G - 0.9G
Upper Frequency	0.5 Hz	5.0 Hz	5.0 Hz
Total Moving Wgt.	3850 lbs.	3850 lbs.	4000 lbs.

Loads on the piston to be encountered under the new performance goals are the static deadweight, 1G acceleration plus deadweight and peak pressure loads. The heave drive hydraulic power supply is capable of producing 500 gal/min at cyclic pressure of up to 1000 psi. Pressure can be limited to normal operating pressures that will produce 1G by adjusting the high pressure relief valves. Maximum compressive loads should be considered at 1000 psi and full extension of the piston.

Compressive loads on the piston are as follows:

Static Deadweight		4000 lbs.
Peak Acceleration (1G) plus deadweight		8000 lbs.
Peak Force at 1000 psi at diameter		
	3.5 inches	9621 lbs.
	3.75 inches	11045 lbs.
	4.00 inches	12566 lbs.
	4.25 inches	14186 lbs.
	4.50 inches	15904 lbs.
	4.75 inches	17720 lbs.
	5.00 inches	19635 lbs.

Euler's equations for slender columns, i.e., length/radius=120, can be used to evaluate the unsupported portion of the heave piston (3). The piston is considered to be a slender ideal column fitted with round ends to which a compressive force P is applied along the principal axis. The critical load P_{cr} is the maximum load which the column can be expected to support

$$P_{cr} = \frac{\Pi^2 \quad EI}{L^2} \qquad (2)$$

where P_{cr} is the critical load

E is the modulus of elasticity

I is the section moment of inertia

L is the unsupported column length

The results are shown for seven diameters and six wall thicknesses at maximum theoretical extension of the piston 28.25 feet.

Wall Thickness

Outside Dia.	.250	.3125	.375	.4375	.500	.625
3.50	8734	10338	11745	12974	14038	15737
3.75	10900	12949	14766	16370	17777	20070
4.00	13398	15967	18267	20315	22132	25144
4.25	16251	19423	22283	24853	27152	31018
4.50	19484	23346	26851	30023	32883	37751
4.75	23120	27765	32006	35867	39372	45404
5.00	27183	32710	37783	42427	46666	54034

Five diameter/wall thickness combinations met the 2:1 safety factor for compression loads (as shown on the previous page). An additional load that the piston must carry is caused by bending due to whip under dynamic conditions, plus the bending due to eccentricity of the applied loads with cabin pitch and roll. The bending stress (4) is given by

$$S_B = \frac{W_{dc}}{I} \qquad (3)$$

where

S_B	is the bending stress psi
W	is the applied load lbs.
d	is the deflection in inches
c	is the distance from the neutral axis to the point of maximum stress
I	is the section moment of inertia

The critical deflection d_{cr} versus column length is determined by subtracting the compression stress, S_c, (applied load/annular area) from the allowable slender column stress, S_{cr} (critical load, P_{cr}/annular area) substitute this for the bending stress S_B and compute the allowable or maximum deflection

$$d_{cr} = \frac{\left(S_{cr} - S_c\right) I}{W_c} \qquad (4)$$

Figure 1 shows the allowable deflection versus column length for three of the five selected pistons. The two other curves fall between the outer curves on the graph.

Previous tests on the MOGEN at the Goleta facility revealed a peak piston deflection of an estimated ± 1/2 inch at a heave drive excitation of 5.3 to 5.4 Hz [5]. This amount of whip was allowable under 1G acceleration loading across the entire working stroke of the original piston, but was not allowable within the highest 8% of the working stroke of the original configuration (± 11 ft.) or the highest 6% of the reduced working stroke (± 10 ft.) operating under 1000 psi driving pressure.

All proposed column configurations had an allowable deflection greater than 1 inch at the maximum working stroke of 26 feet and 1000 psi driving pressure. Operating loads of 8000 lbs. (deadweight of 4000 lbs. + 1G) would further raise the allowable deflection.

Another method of assessing a column's stiffness would be to calculate its natural frequency. Assuming a uniform circular hollow beam of varying length and hinged ends, the following formula applies

ALLOWABLE DEFLECTIONS
VS
COLUMN LENGTH
at maximum loads

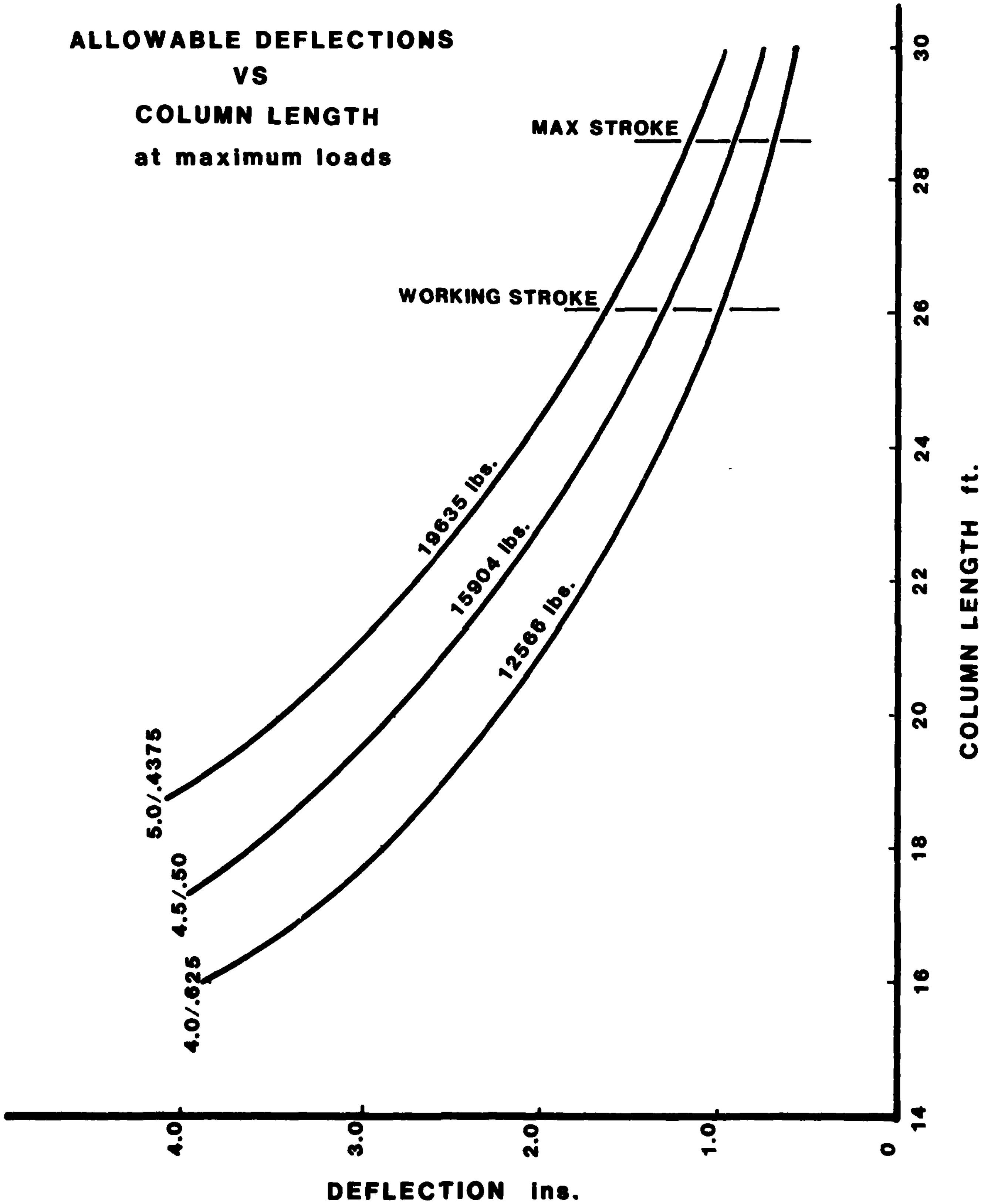


Figure 1

$$f = \frac{A}{2\Pi}\left[\frac{E\ I}{\mu\ L^4}\right]^{1/2} \tag{5}$$

where f is the natural frequency

A is the end conditions (9.87 for hinged ends)

E is the modulus of elasticity (psi)

I is the moment of inertia ($in.^4$)

μ is the mass per unit length of beam, $\frac{lb.-sec.^2}{in.^2}$

and L is the length of beam (in.)

Figure 2 shows that three columns (4.5/.50, 4.75/.4375, 5.00/.4375) have natural frequencies that meet or exceed that of the original column.

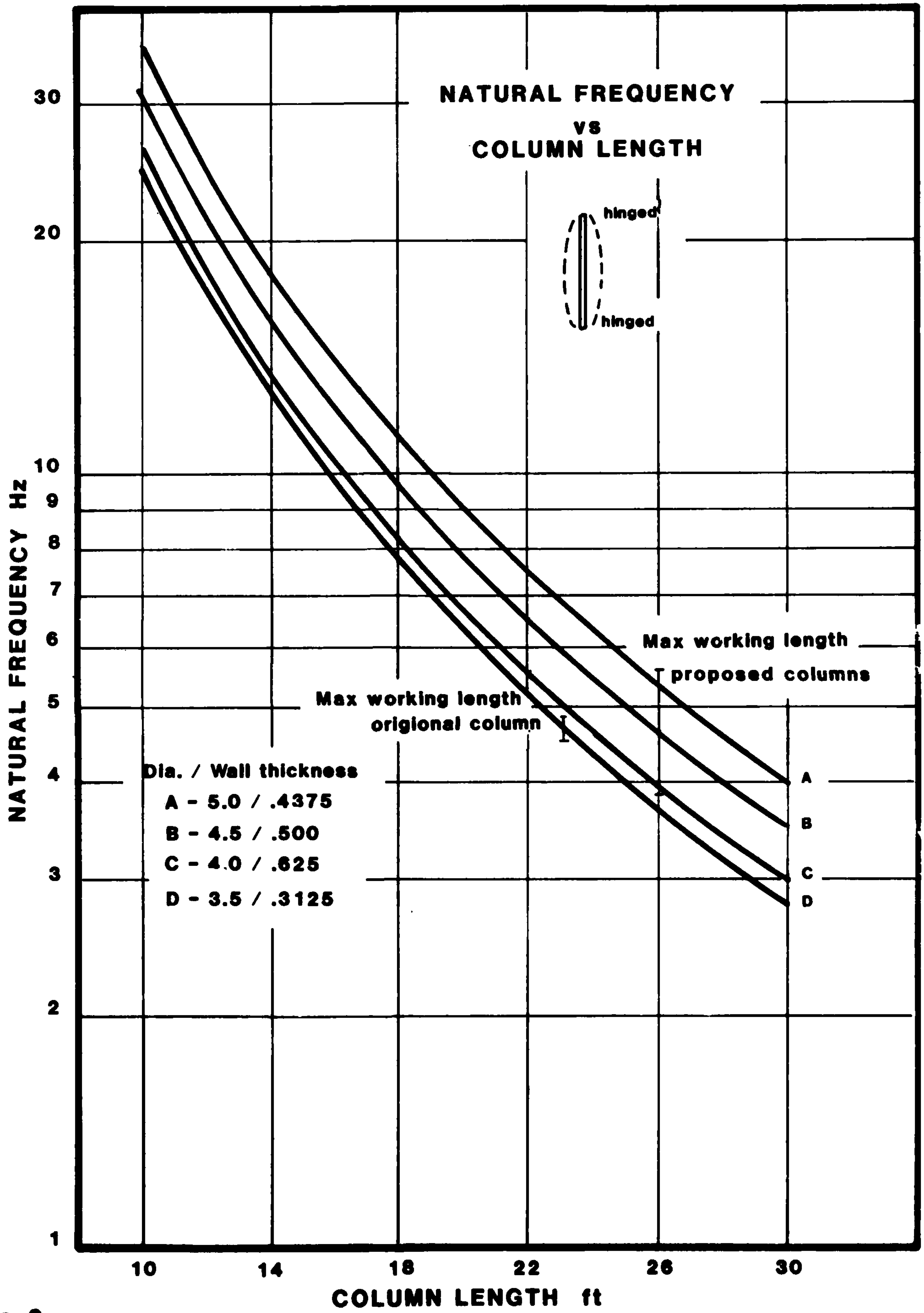
As piston diameters increase, pressures required to produce the desired acceleration levels decrease, but in order to obtain the desired velocity levels, the flow rate must increase and it is always easier and less costly to decrease pressure than it is to increase flow rates. With this in mind it was decided to choose the smallest diameter piston that would meet all the criteria. Using the formula

$$\text{Flow} = \text{Piston Area} \times \text{Piston Velocity} \tag{6}$$

and knowing that the upgraded MOGEN could achieve 17 ft./sec. then the flow rate of the upgraded heave drive hydraulic power supply is 510 GPM. With the same formula the velocity for each of the candidate pistons was found along with the flow rate required for a maximum velocity of 17 ft./sec. Maximum pump pressure can also be calculated for 1G upward acceleration and 4000 lb. static deadweight with the formula

$$\text{Pressure} = \frac{\text{Deadweight} + 1G}{\text{Piston Area}} \tag{7}$$

Piston Diameter (in.)	Piston Velocity @510 GP	Flow Rate GPM @ 17 fps	Pump Pressure w/4000 lbs. and 1G Acc.
3.5	17.0	510	859
4.0	13.0	666	637
4.25	11.5	751	564
4.50	10.3	843	503
4.75	9.2	939	451
5.00	8.3	1040	407



NATURAL FREQUENCY
vs
COLUMN LENGTH
hinged
hinged
Max working length
proposed columns
Max working length
origional column
Dia. / Wall thickness
A - 5.0 / .4375
B - 4.5 / .500
C - 4.0 / .625
D - 3.5 / .3125
A
B
C
D
NATURAL FREQUENCY Hz
30
20
10
9
8
7
6
5
4
3
2
1
10
14
18
22
26
30
COLUMN LENGTH ft

Figure 2

The 4.5 inch diameter column with a 0.5 inch wall thickness is the smallest diameter that met all the criteria of a 2:1 safety factor.

The physical parameters for the recommended piston are:

Outside diameter	4.5 inches
Inside diameter	3.5 inches
Wall thickness	0.5 inches
Working Stroke	22 feet
Unsupported length at bottom of stroke	48 inches
Unsupported length at top of stroke	26 feet
Unsupported length at top of buffer	28.25 feet
Total length	29.25 feet
Piston cross sectional area	15.90 inches^2
Piston annular area	6.28 inches^2
Radius of Gyrations	1.15 inches
Moment of Inertia	12.763 inches^4
Weight	597 pounds

HEAVE DRIVE POWER SUPPLY

With the piston selected to meet the structural, safety, and operational requirements, the hydraulic power supply, lines, and valves were sized to compliment the piston. As can be seen from the table on page 7, the 4.5 inch diameter piston will require 503 psi to meet the + 1G acceleration and 843 GPM to achieve + 17 ft./sec. The heave drive power supply used with the upgraded MOGEN was more than adequate to meet the pressure requirements but had only 60 percent of the flow capacity needed for the present system. Specifications for the power supply, built by ACL-Filco Corporation, are as follows:

Pressure - 500 psig normal, 1000 psig cyclic

Temperature - 100^{o}F. normal, 130^{o}F. shutdown

Reservoir Capacity - 1000 gallon

Pumps

Cooling Pump	250 gpm @ 50 psig
Power Pumps (2 each)	250 gpm @ 500 psi and 150 SSU

A hydraulic power supply identical in pressure and capacity to the upgraded one was purchased and installed in parallel. For reasons of economy, maintenance, and performance, it was of paramount importance that the power pumps be exactly the same make and model as those presently used, i.e. DeLavel IMO Pump, Type A6DH-400. This configuration allows the pumps to be operated independently during reduced operational requirements.

No. of Pumps	1	2	3	4
Max. Vel. (fps)	5.0	10.1	15.1	20.2
Max. Dist. (ft.)	11.0	11.0	11.0	11.0
Acc. @ .05 Hz & X max. (G)	.0338	.0338	.0338	.0338
Freq. @ V max. & X max. (Hz)	.0732	.1461	.2185	
Acc. @ V max. & X max. (G)	.0708	.2887	.6453	
Freq. @ 1G				.2720
Acc. @ 1 Hz	.978			
Vel. @ 1G				18.79

Figure 3 was generated from the above chart showing the theoretical performance limits with an increasing number of pumps running at ± 11 feet for the 4.5 inch-diameter piston. The system is velocity limited until all pumps are on with the larger piston.

SUPPLY AND RETURN VALVES, FITTINGS, AND PIPES

The heave drive system was designed to accommodate the larger piston with the original stroke. All new larger supply and return lines were installed because of the increased flow.

With a doubling of the pumping capacity the hydraulic pipes and valves were increased to accommodate the necessary hydraulic fluid displacements with acceptable fluid velocities. Industry standards for supply line velocities are 7 to 15 ft./sec. and return line velocities of 2 to 4 ft./sec. Using the formula

$$D = \left[.4085 \ \frac{Q}{V} \right]^{1/2} \tag{8}$$

PISTON DIAMETER 4.5"

DISPLACEMENT ± 11.0'

ACCELERATION 1G MAX.

NUMBER OF PUMPS	1	2	3	4
VELOCITY (fps)	5.0	10.1	15.1	20.2

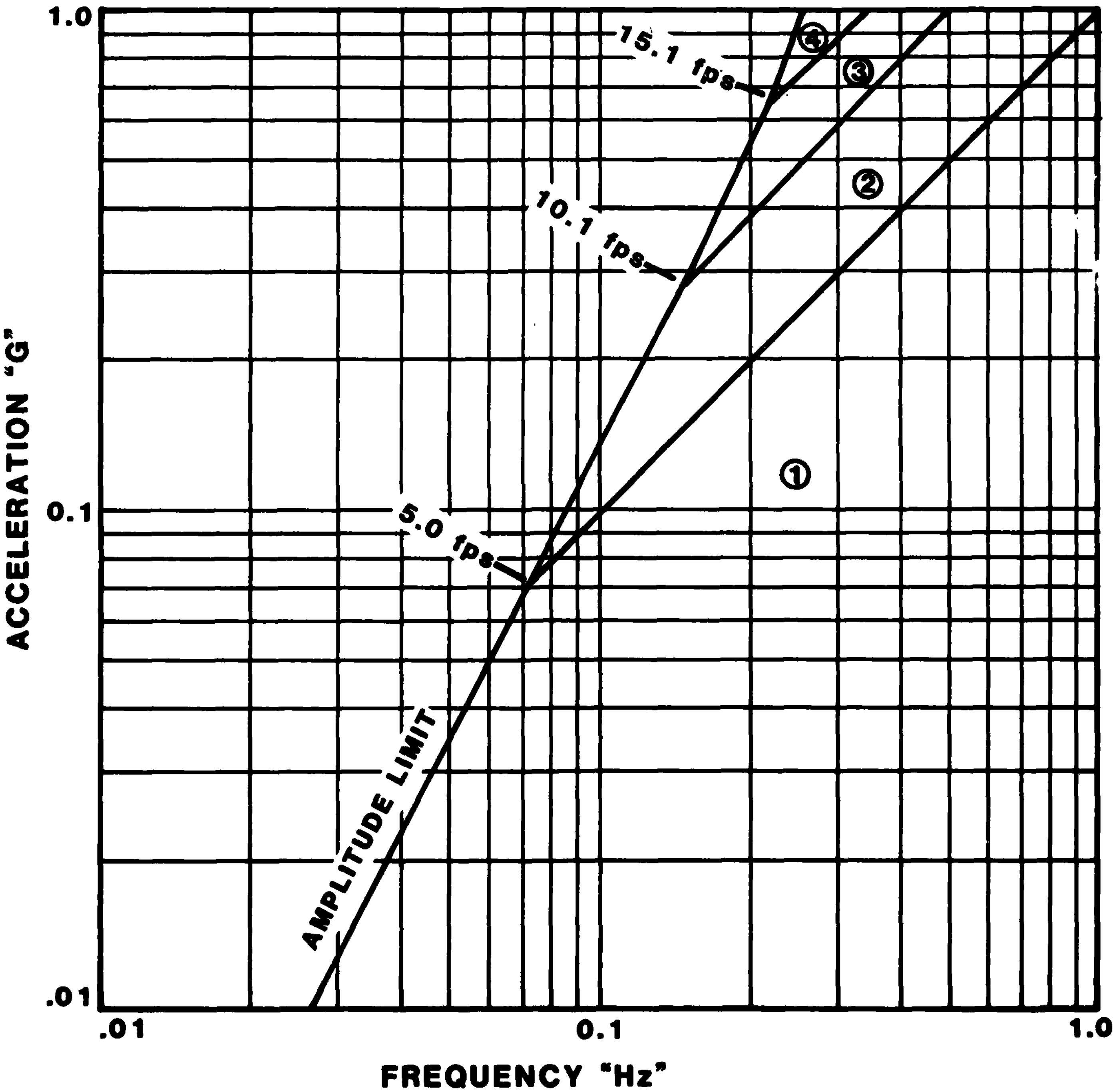


FIG.3 HEAVE OPERATING REGION

where

D is the internal pipe diameter inches

Q is the flow in GPM

V is the fluid velocity in ft./sec.

.4085 includes all the conversion factors.

Using 10 fps and 1000 GPM as the supply velocity and flow (the calculations for the original upgrade used 15.3 ft./sec.) the pipe internal diameters computes to be 6.39 inches. With the nearest standard pipe diameter being 6.00 inches, the oil velocity is a very acceptable 11.3 ft./sec.

Repeating the calculations for the return line with 3 ft./sec. and 2000 GPM (flow from pumps plus flow from the cylinder), the internal diameter comes out to be 16.5 inches. This is unacceptably large considering the transition pieces, bends, connections, and trench requirements. Eight-inch diameter Schedule 10 pipe, with a 8.124 ID is readily available and has a velocity of 12.4 ft./sec. at maximum flow. This velocity is higher than recommended, but with the short run of pipe the pressure drop is minimal.

The traverse stress due to internal pressure on the supply line (there is no pressure on the return line) is a hoop tension that is found from the relation

$$S_1 = \frac{p\,d}{2\,t} \tag{9}$$

where

S_1 is the hoop stress

p is the internal pressure

d is the inside diameter

and t is the wall thickness.

Selecting Schedule 40 pipe and maximum pressure of 1000 psi, the stress on the 6-inch pipe is 10830 psi. Allowable stress for seamless pipe in oil piping systems, within refinery limits, is 17450 psi for temperatures up to 150°F. [6]. With a yield point of 3000 psi this gives a safety factor of nearly 2.8 for the supply line. The same formula gives a stress of 12422 psi for the same conditions on the 8-inch heave drive casing. Even with all the above calculations, Schedule 80 piping was installed to dispel any doubts about the strength of the piping.

Obtaining a 1G acceleration in the positive Z direction will be no problem as the heave drive pumps develop more than enough pressure, but to obtain a negative 1G acceleration would require zero back pressure through the valves and piping. As this was not possible, the next best solution was to minimize the pressure drop through the valves and fittings.

The 6-inch safety valve has a C_v of 944 when fully open at maximum pressure and flow, and the 6-inch control valve is rated at a C_v of 394 [7], [8]. With the relation

$$\Delta p = S\left(\frac{Q}{CV}\right)^2 \qquad (10)$$

where

Δp is the pressure drop in psi

S is the specific gravity

Q is the flow in GPM

and

C_v is the flow coefficient

and using .871 for the specific gravity of Tellus 32 oil and a Q of 1000 and 2000 GPM, respectively. The pressure drop for the safety valve is .97 psi and 22.44 for the control valve.

NOTE: The flow coefficient C_v is defined as the number of US gallons per minute of water at 70°F., which will flow through a valve at a pressure drop of 1 psi.

A schematic for the heave drive hydraulic system is shown in figure 4. An equivalent length of pipe can be obtained for the fittings using the relation

$$P_\lambda = \frac{.000668\ Z\ LV}{d^2} \qquad (11)$$

$$\text{and } Z = \mu S \qquad (12)$$

where

P is the pressure drop in psi in L feet of pipe

z is the kinematic viscosity in centipoise

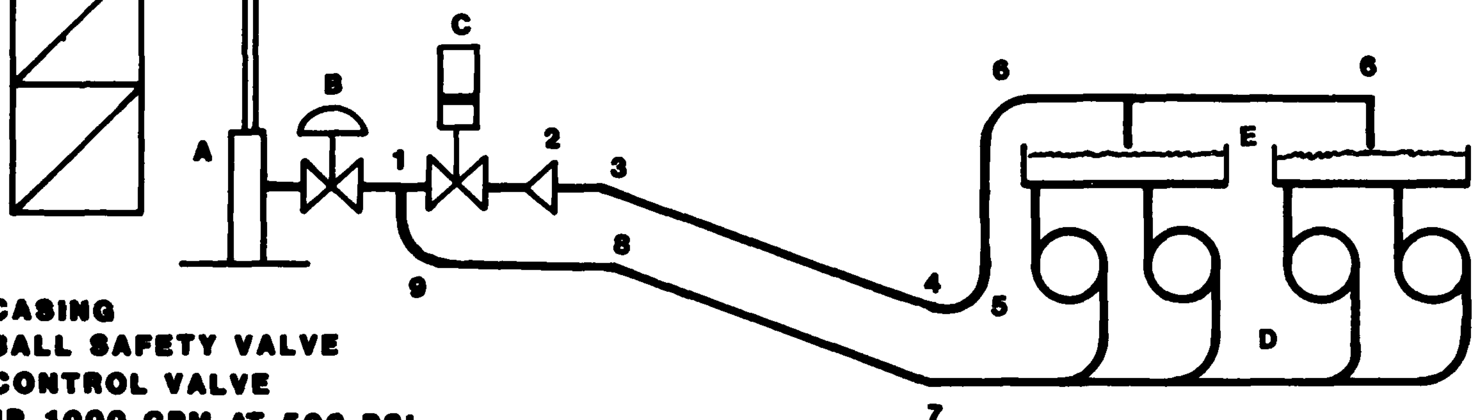
L is the equivalent length of pipe in feet = 114.6

v is the velocity in ft./sec. = 12.4

d is the pipe I.D. in inches = 8.125

μ is the dynamic viscosity in centistokes = 32

S is the specific gravity = .8708



RETURN LINE

EQUIVALENT LENGTH OF PIPE - FT

1	THROUGH 6" TEE	6.75
2	REDUCER 6"x 8"	-
3	21° BEND R/d - 2 8"	6.98
4	45° ELL R/d - 2 8"	6.98
5	90° ELL R/d - 2 8"	5.98
6	90° ELL R/d - 2 8"x 2	11.98
LENGTH	2 - 3	14.0
LENGTH	3 - 4	30.0
LENGTH	4 - 5	21.0
LENGTH	5 - 6	11.0
		114.63 FT

A 8" CASING
B 6" BALL SAFETY VALVE
C 6" CONTROL VALVE
D PUMP 1000 GPM AT 500 PSI
E 2000 GAL RESERVOIR
7 45° ELL R/d - 2 6"dia
8 21° BEND R/d - 2 6"dia
9 90° BEND R/d - 2 6"dia

Figure 4

The pressure drop in the return line is 0.40 psi.

Adding the back pressure from the safety valve, the control valve, and the 8-inch return pipe, a minimum pressure of 23.81 psi is obtained.

Rearranging Newton's formula, F = ma and substituting pressure x area for force, the maximum cabin deceleration is found to be 0.91G.

$$G = 1 - \frac{PA}{W} \tag{13}$$

where

1 is the free fall acceleration in G's

p is the back pressure = 23.81 psi

A is the piston area = 15.9 inches^2

W is the moving weight = 4000 pounds

G is the acceleration = in G's

It should be noted that both an increase in back pressure and a decrease in moving weight will result in a slower deceleration.

CONCLUSIONS

Returning the MOGEN to its original heave displacement while retaining the upgraded velocity requirement with the increased safety factor, represented a major redesign effort. The only feasible time to accomplish this work was while the device was disassembled and the new facility was being designed.

The modifications satisfied all the requirements as well as allowing for some future increase in performance in stroke, velocity, and acceleration without a major expenditure.

REFERENCES

1. Alfred Z. Boyajian and Wilton A. Stewart, "Upgraded Motion Generator Structural Integrity, Performance and Operating Safety," Coby Associates Report 723.139 dated June 30, 1975. Coby Associates, P. O. Box 158, Pacific Palisades, CA 90272. Page 78.

2. Ibid. Pages 117-118.

3. F. E. Miller and H. A. Doeringfeld, "Mechanics of Materials, " 1955, International Textbook Company. Page 368.

4. Ibid. Page 145.

5. Alfred Z. Boyzjian and Wilton A. Stewart, "Upgraded Motion Generator Structural Integrity, Performance and Operating Safety," Coby Associates Report 723.139 dated June 30, 1975. Coby Associates, P. O. Box 158, Pacific Palisades, CA 90272. Page 121.

6. Sabin Crocker, "Piping Handbook," 1945, page 1156. McGraw-Hill Book Company, Inc.

7. Fisher Controls Bulletin PS Sheet Catalog 10, page 1-134:A, dated October 1, 1977/DEH.

8. Fisher Controls Catalog 10, pages 1-78, dated May 1974.

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